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ABSTRACT

This paper numerically and experimentally analyzed mechanical loss of an eccentric bearing to improve mechanical efficiency of a rotary compressor. On the basis of dynamic analysis on eccentric bearing, the dynamic load of the eccentricity bearing was solved, and the lubrication of eccentric bearing was analyzed. The trajectory of the bearing was calculated to analyze the fluid dynamic lubrication of the bearing in order to find out which parts may be wear. The numerical results show that an eccentric bearing with cut off frictional surface gives higher mechanical efficiency than that of a normal eccentric bearing. The experimental results showed increase in coefficient of performance (COP) by 1% which correspond to the numerical results. In addition, to confirm the reliability of an eccentric bearing with cut off frictional surface, we did the reliability test under conditions of optimum shape. This paper concludes that an eccentric bearing with an optimum shape is effective for obtaining higher efficiency.

1. INTRODUCTION

Due to recent environment problems represented by global warming, a demand for electrical appliances that have greater energy saving properties and less impact on the environment such as air conditioners which occupy highest percentage among electric consumed in China has been increasing. It is therefore important to develop a highly efficient rotary compressor to reduce the power consumption of air conditioners. In order to improve an efficiency of a rotary compressor, it is necessary to reduce the friction loss of the sliding surfaces. There are many sliding surfaces in a rotary compressor such as the surfaces between vane and roller, shaft and bearings, roller and flange, crank and roller. In this paper, we focused on the friction loss of an eccentric bearing (consists of crank and roller) which has highest percentage among the friction loss as shown in Figure 1. On the basis of dynamic analysis on eccentric bearing, the dynamic load of the eccentricity bearing was solved, and the lubrication of eccentric bearing was analyzed. And it was found that it is effective to cut off the frictional surface.

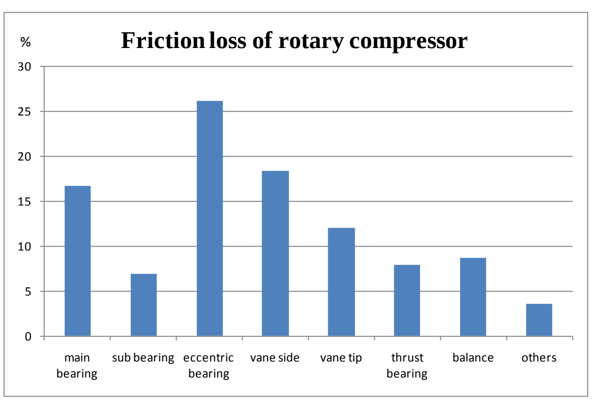


Figure 1 the friction loss of rotary compressor under standard condition (Calculated)

2. NUMERICAL ANALYSIS

In a rotary compressor, an eccentric bearing consists of crank and roller. We developed a program which has analyzed mechanical loss and minimum oil film thickness of an eccentric bearing under various conditions.

2.1 Bearing load

Figure 2 shows the forces acting on the roller. They are gas force (F_g), inertia force (F_e), the contact forces between roller and vane tip (F_{vn} and F_{vt}), the friction moment on both end surface of roller (M_a), the friction moment due to the oil film viscosity between roller and crank (M_{eb}), and the forces due to the refrigerant viscosity between roller and cylinder (F_{cn} and F_{ct}).
The bearing load F_{en} is as follows:

$$F_{en} = \sqrt{F_{rx}^2 + F_{ry}^2} \qquad (1)$$

$$F_{rx} = F_g \cos(\frac{\theta - \alpha}{2}) + F_a \sin\theta + F_e \cos\theta - F_{vn} \cos\alpha - F_{vt} \sin\alpha - F_{cn} \cos\theta + F_{ct} \sin\theta$$

$$F_{ry} = F_g \sin(\frac{\theta - \alpha}{2}) - F_a \cos\theta + F_e \sin\theta + F_{vn} \sin\alpha - F_{vt} \cos\alpha - F_{cn} \sin\theta - F_{ct} \cos\theta$$

$$\eta_{eb} = tg^{-1} \frac{F_{ry}}{F_{rx}} \qquad (2)$$

And the motion equation of the roller is shown as follows:

$$I_r \ddot{\omega}_r = M_{eb} - M_a - R_{ro} * (F_{vt} + F_{ct}) \qquad (3)$$

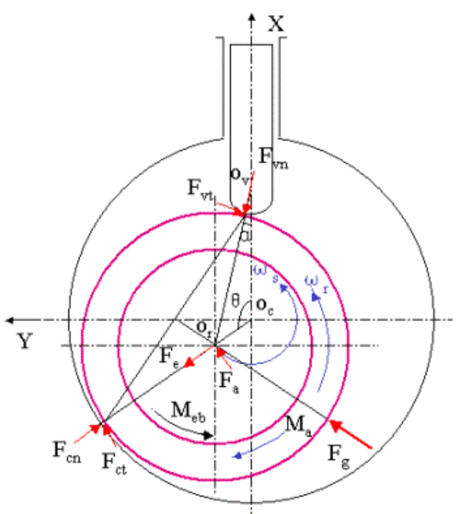


Figure 1 Analytical model of roller

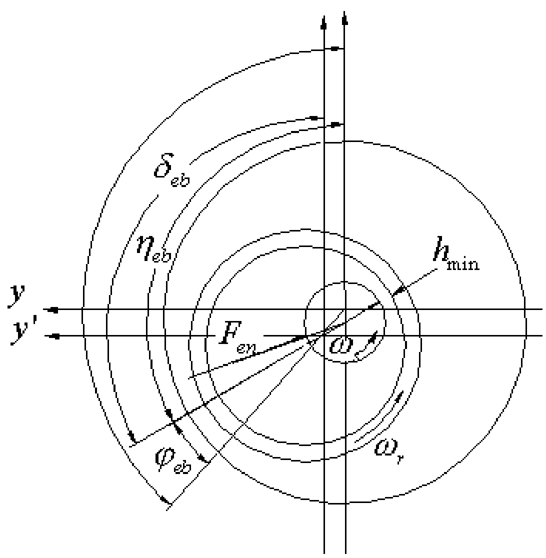


Figure 2 Eccentric bearing model

2.2 Friction moment of normal eccentric bearing

The minimum film thickness also directly affects the conventional lubrication regime determination using the parameter "Λ" (equation 2). Λ is typically used to determine the lubrication regimes that reasonably well demonstrate the contacting surfaces, which directly influences the wear behavior of a particular lubricated system. Typically, the lubrication regimes, by using the Λ parameter, are used as follows. $\Lambda \geqslant 3$ for hydrodynamic lubrication or elastohydrodynamic lubrication, $3 \geqslant \Lambda \geqslant 1$ for mixed lubrication and $1 \geqslant \Lambda$ for boundary lubrication regime, which is schematically presented in Figure 3. a more precise classification also includes $\Lambda \geqslant 4$ for fully elastohydrodynamic lubrication with no effects on wear, and mixed lubrication regime is also divided up from 1 to 1.5 and from 1.5 to 3, indicating the different influences on wear and lifetime (Kalin and Velkavrh, 2009) (Stachowiak and Batchelor,2005) (Tallian, 1967).

$$\Lambda = \frac{h_{\min en}}{\left[h_{\min}\right]_{eb}} = \frac{h_{\min en}}{\sqrt{R_{zeb1}{}^{2} + R_{Zeb2}{}^{2}}} \qquad (4)$$

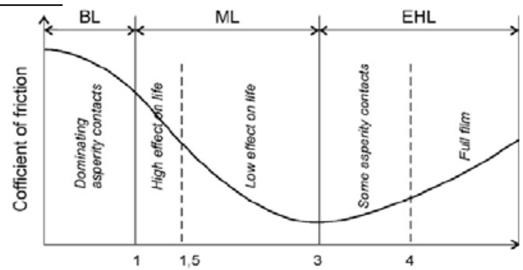


Figure 3 A schematic of a typical classification of different lubrication regimes depending on Λ parameter

The friction moment M_{eb} on the inside surface of roller by crank is calculated by the attitude ε_{eb} and attitude angle of eccentric journal bearing δ_{eb}. Since magnitude and direction of bearing load are great, the variations of ε_{eb} and oil film thickness of the bearing are also great. If the oil thickness is less than a certain value at some rotating angle the bearing is in boundary lubrication condition and the frictional torque is as follows:

$$\mathrm{M}_{eb} = \begin{cases} \dfrac{2\pi\mu_{oeb}B_{eb}R_{eb}{}^{3}(\omega_s - \omega_r)}{C_{eb}\sqrt{1-\varepsilon_{eb}^2}} - \dfrac{1}{2}F_{eb}C_{eb}\varepsilon_{eb}\sin(\delta_{eb} - \eta_{eb}) \quad , \qquad \Lambda \geq 1 \\ \mu_{eb}F_{eb}R_{eb}\cos(\delta_{eb} - \eta_{eb}) \quad , \qquad \Lambda < 1 \end{cases} \qquad (5)$$

The attitude ε_{eb} and attitude angle of eccentric journal bearing δ_{eb} are given from the differential equation of loci shaft center of the bearing:

$$\dot{\varepsilon}_{eb} = \frac{F_{en}\psi_{eb}{}^{2}}{B_{eb}D_{eb}S_{os}\mu_{oeb}}\left[\cos(\delta_{eb} - \eta_{eb}) - \frac{\sin\left|\delta_{eb} - \eta_{eb}\right|}{tg\beta_r}\right] \qquad (6)$$

$$\dot{\delta}_{eb} = \frac{1}{2}\left[\omega_r + \omega_s - \frac{\sin(\delta_{eb} - \eta_{eb})}{\sin\beta_r}\frac{F_{en}\psi_{eb}{}^{2}}{B_{eb}D_{eb}S_{or}\mu_{oeb}}\right] \qquad (7)$$

2.3 Friction moment of new design eccentric bearing

In order to reduce the friction loss of eccentric bearing, we developed an eccentric bearing with cut off frictional surface as shown Figure 9-10. Therefore, the following equation gives friction moment M_{eb} for it,

$$\mathrm{M}_{eb} = \begin{cases} \dfrac{2\pi\mu_{oeb}B_{eb}R_{eb}{}^{3}(\omega_s - \omega_r)}{C_{eb}\sqrt{1-\varepsilon_{eb}^2}}[\mathrm{A}(\beta 1+2\pi)\text{-}\mathrm{A}(\beta 2)] - \dfrac{1}{2}F_{eb}C_{eb}\varepsilon_{eb}\sin(\delta_{eb} - \eta_{eb}) \quad , \qquad \Lambda \geq 1 \\ \mu_{eb}F_{eb}R_{eb}\cos(\delta_{eb} - \eta_{eb}) \quad , \qquad \Lambda < 1 \end{cases} \qquad (8)$$

$$\mathrm{A}(\theta) = \tan^{-1}\left(\sqrt{\frac{1-\varepsilon_{eb}}{1+\varepsilon_{eb}}} * \tan\frac{\theta}{2}\right)$$

3. NUMERICAL AND EXPERMENT RESULTS

3.1 Numerical results

Figure 4-5 show the numerical results of the bearing load and the minimum oil film thickness of normal eccentric bearing under standard operating condition as shown in Table 1. The peak of the bearing load is about 1500N when the shaft angle is about 210 °. And the minimum oil film thickness is about 2μm, the bearing is in mix lubrication condition because the Λ parameter is about 1.12. And Figure 6 shows the trajectory of the bearing. it is found that the direction does not rotate but remain within a constant range (about from 275° to 40°) .

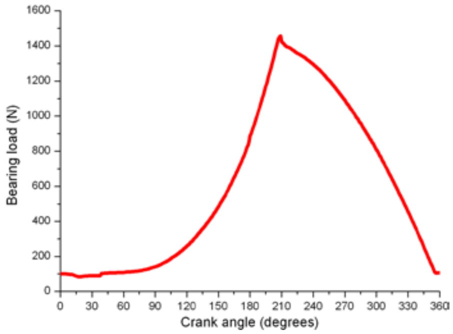


Figure 4 Bearing Load

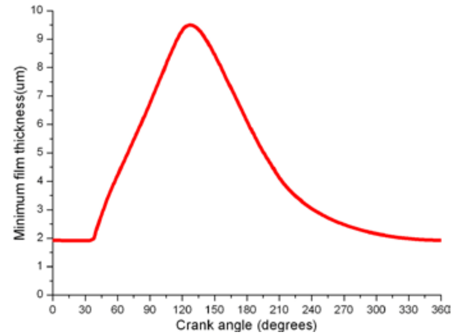


Figure 5 Minimum film thickness of eccentric bearing

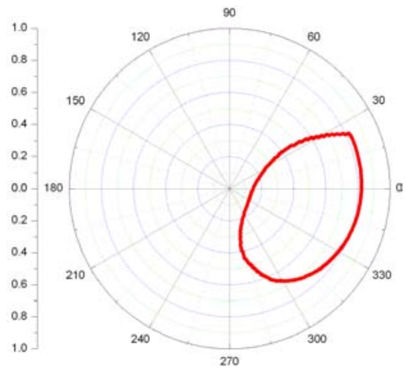


Figure 6 Bearing trajectory

Table 1 Standard condition in simulation

Item	Values	Unit
Suction pressure, P_s	1.084	MPa
Discharge pressure, P_d	2.765	MPa
Rotating frequency, f	60	Hz

Figure 7-8 show the numerical results of the minimum oil film thickness and the trajectory of normal eccentric bearing under heavy load condition as shown in Table 2. The minimum oil film thickness is about 1μm, the bearing is in boundary lubrication condition because the Λ parameter is about 0.56. And the direction remain within a constant range (about from 300° to 25°) .

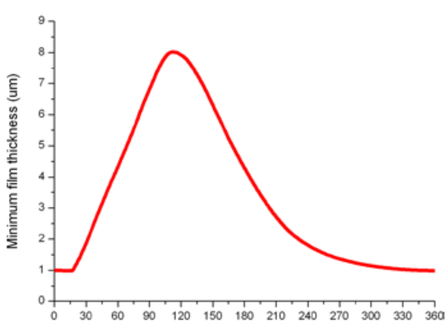


Figure 7 Minimum film thickness of eccentric bearing

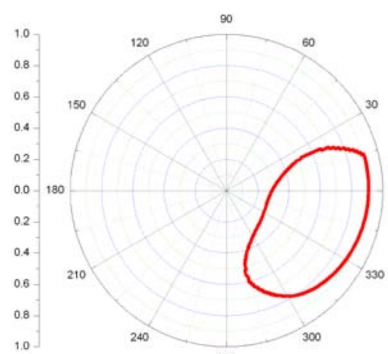
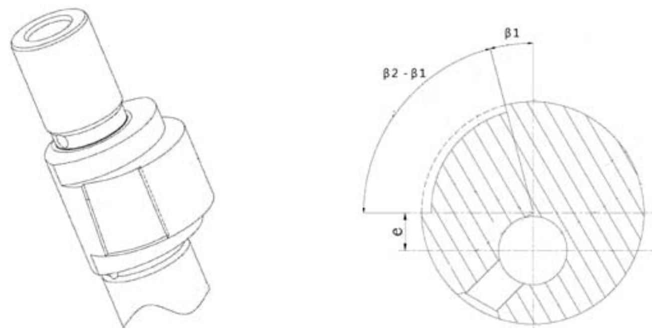


Figure 8 Bearing trajectory

Table 2 Heavy load condition in simulation

Item	Values	Unit
Suction pressure, P_s	1.15	MPa
Discharge pressure, P_d	4.25	MPa
Rotating frequency, f	60	Hz

Base on the above numerical results of normal eccentric bearing, we design two eccentric bearing models with cut off frictional surface as show in Figure 9 and Table 3. The area of the cut off frictional surface of model A is more than that of model B. And the numerical results under standard condition are shown in Table 4.



(a) (b)

Figure 9 Scheme of New eccentric bearing

Table 3 New eccentric bearing model

Model	β 1	β 2	Cut off area (β 2- β 1)
model A	10°	130°	120°
model B	60°	140°	80°

Table 4 Numerical results under standard condition

Model	Capacity	Power consumption	COP
normal eccentric bearing	100%	100%	100%
model A	100%	98.7%	101.2%
model B	100%	99%	101%

3.2 Performance test results

We conducted experiment of a rotary compressor by varying bearing shapes of crankshaft. Table 5 shows the experiment results under standard condition. The results show that the new models give better efficiency than that of a normal eccentric bearing. It is also found that the experimental results showed increase in coefficient of performance (COP) by 1% which correspond to the numerical results.

Table 5 Experiment results under standard condition

Model	Capacity	Power consumption	COP
normal eccentric bearing	100%	100%	100%
model A	100.1%	98.8%	101.3%
model B	99.9%	98.9%	101%

3.3 Reliability test results

In addition, to confirm the reliability of an eccentric bearing with cut off frictional surface, we did the reliability test under heavy load condition. After 500 hours test, wear took place in the crank of Model A as shown Figure 10. But it did not happen in model B especially after 1500 hours test as shown in Figure 11. Because the bearing is in boundary lubrication condition and the direction remain within a constant range (about from 300° to 25°) under the heavy load condition as shown in Figure 8. And the cut off frictional surface of model A came into the range, made the oil film pressure decreasing, the thickness thinner. But model B did not come into the range, so it was better.

Figure 10 Model A (500 Hours under heavy load condition)

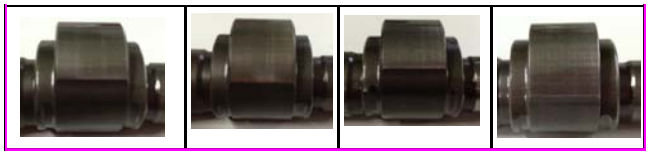


Figure 11 Model B (1500 Hours under heavy load condition)

4. CONCLUSIONS

Numerical and experimental analyses show that the new models give better efficiency than that of a normal eccentric bearing. It is also found that the experimental results showed increase in COP by 1% which correspond to the numerical results. In addition, to confirm the reliability of an eccentric bearing with cut off frictional surface, we did the reliability test under conditions of optimum shape. This paper concludes that an eccentric bearing with an optimum shape is effective for obtaining higher efficiency.

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